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THE EFFECT OF INLET CONDITIONS ON THE FLOW IN ANNULAR DIFFUSERS

by

I.H.JOHNSTON

JANUARY, 1953

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NATIONAL GAS TURBINE ESTABLISHMENT

The effect of inlet conditions on the flow in annular diffusers

- by -

I. H. Johnston

SUMMARY

Tests have been carried out on annular diffusers having a common area ratio of 3.19 and varying in divergence angle from 6.5° to 15°. The performance of each diffuser has been measured for a variety of inlet velocity distributions and the effect of axially splitting the flow in the diffusers has been investigated.

Diffuser efficiency is found to deteriorate as inlet conditions become non-uniform, this tendency increasing with diffuser angle.

Splitting of the higher angle diffusers improves efficiency for non-uniform profiles, but these increases in efficiency are accompanied by pronounced static pressure gradients across the diffuser throat which in certain applications might prove undesirable.

CONTENTS

ILLUSTRATIONS

1.0 Introduction

The tests described in this report represent an extension of the work reported by Ainley in Ref. 1. The latter paper gives values of efficiency of pressure recovery for a group of annular diffusers all having the same area ratio but with varying angles of divergence, and describes the effect of introducing axial splitters along the length of these diffusers. These tests were all made with a uniform radial distribution of velocity at entry to the diffusers, but as this ideal condition seldom exists in practice a further investigation has been made to determine the effect of some non-uniformities of inlet velocity profile on diffuser efficiency.

It should be noted that only radial variations have been considered, (i.e., the circumferential distributions have been kept uniform) and that only a few of the infinite number of possible distributions have been investigated.

2.0 Apparatus

The diffuser rig incorporating an 8.5° diffuser is shown in Figure 1. The assembly was composed of three parts; an inlet section in which the air was accelerated into an annulus of 10 in. O.D. and 7½ in. I.D., a conical diffusion section (having in the assembly illustrated in Figure 1 an included angle of 8.5°), and a parallel outlet section of 12¾ in. O.D. and 4¾ in. I.D. The 8.5° diffuser section was interchangeable with three other diffusers all having the same area ratio, namely 3.19:1, but varying in length to give included angles of 6.5°, 10.5° and 15° respectively. Mean diameter splitters were provided for all except the 6.5° diffuser. As indicated in Figure 1 each splitter took the form of a cylindrical plastic shell situated at the mean diameter of the air path and located by four spiders. When in position the splitter projected about 1 in. upstream of the diffuser inlet and both splitter and spiders extended along the entire diffuser length. Air was supplied to the inlet from a 30 H.P. fan via a length of straight ducting containing an orifice plate which provided for measurement of mass flow.

The inlet velocity profile was varied by introducing gauzes at a convenient flange about 8 in. upstream of the diffuser throat. Static pressure tappings were located at the inlet in both the inner and outer walls and also in the outer wall immediately after the junction between diffuser and outlet section. All pressures were taken to be the mean of four tappings spaced equally round the annulus. An L-shaped radially traversing pitot was positioned in the plane of the inlet statics and was used in determining total head distributions at entry to the diffusers. All the components of the rig were made of wood and the surfaces adjacent to the air flow had a smooth polished finish.

3.0 Description of tests

Six gauze arrangements were used to give various distributions of total head at inlet to the diffusers and both plain and split diffusers were tested with each arrangement. The first tests were performed with a single wide-meshed gauze extending across the entire flow path. This gauze gave a uniform inlet total head distribution and also acted as a support for other gauzes which in subsequent arrangements extended only partially across the annulus. For each test, readings of mass flow, temperature and static pressure were taken and the inlet total head distribution was traversed in steps of 1/10 in. with the pitot tube.

4.0 Diffuser efficiency

It has been shown (Ref. 2) that from the fundamental energy standpoint the efficiency of a diffuser can be expressed in the form

$$\eta_o = \frac{(p_2 - p_1)\, \bar{u}_1 A_1}{\int_{A_1} \frac{1}{2}\rho u^3 dA - \int_{A_2} \frac{1}{2}\rho u^3 dA} \qquad \ldots\ldots\ldots\ldots (1)$$

where p_2 = outlet static pressure

p_1 = inlet static pressure

$\bar{u}_1$ = mean inlet velocity

A_1 = inlet area

A_2 = outlet area

and where the absolute fluid velocities at entry and exit are assumed to be axial.

By further assumptions of uniform velocity at entry and exit, it can be shown that for incompressible flow the above expression can be simplified to

$$\eta = \frac{p_2 - p_1}{\frac{1}{2}\rho\bar{u}_1^2 \left(1 - A_1^2/A_2^2 \right)} \qquad \ldots\ldots\ldots\ldots (2)$$

Although the assumptions regarding velocity are seldom, if ever, satisfied in the practical case, this latter value for efficiency is in general use and provides a reasonable basis for application of test results to diffuser design. The absolute value η_o is not favoured for the reason that the aim of any diffuser is the conversion of kinetic energy to pressure and thus any kinetic energy over and above the minimum required for uniform flow at outlet should be regarded as an energy loss. All values for efficiency quoted in this paper are therefore based on equation 2.

5.0 Test results

5.1 Accuracy of results

For the plain diffuser tests errors in observation should represent less than ± 1.00% of efficiency, but for the split diffusers the method of calculation employed reduces the accuracy of the results and possible errors may amount to ± 1.50%.

As all tests were carried out with an inlet Mach number of 0.15 the assumption of incompressible flow introduces negligible error.

The Reynolds number of the tests based on the value $R_e = \frac{\rho V_1}{\mu} \times$ (O.D. - I.D.) was 2.5×10^5 compared with design conditions in a typical engine at altitude which give Reynolds numbers of between 4 and 5×10^5 at entry to diffusing sections.

Despite this discrepancy in Reynolds number the conclusions regarding variation of efficiency with inlet profile based on the test results should form a guide to estimating diffuser performance under the conditions obtaining in practice.

5.2 Tests on plain diffusers

In the first group of tests the four diffusers were tested for the three inlet velocity profiles shown in Figure 2. These profiles had values of $V_1 \max/\bar{V}$ of 1.0, 1.15 and 1.30, this peak velocity occurring at approximately mean diameter. The results, shown in Figure 2, indicate a decrease of η with increase of diffuser angle θ, and a further drop in η with increase in $V_{max}/\bar{V}$. Included in Figure 2 are test results taken from Reference 1 for a uniform profile. It can be seen that at the higher angles there is considerable discrepancy between the efficiencies obtained with this profile and with profile (1). This may be attributable to the difference in boundary layer thickness, as at high values of θ the thicker boundary layer will more readily separate under the action of the adverse pressure gradient.

In the second group of tests the peak velocity of the inlet distribution was located at three radial positions, namely O.D., M.D. and I.D. Velocity profiles are shown in Figure 3 together with curves of η v θ. For V_{max} at O.D. the efficiency curve is similar to that obtained with a uniform profile but with the efficiencies lowered by between 1% and 2%. When the peak velocity occurs at M.D. the efficiency curve becomes much steeper, giving the high value of 88% for θ of 6.5° but only 61% when θ increases to 15°. The final profile, number (6), brings V_{max} towards the inner wall and gives low efficiencies at all values of θ, with a minimum value of 42% when θ equals 15°.

Some standard of comparison can be obtained by consideration of the theoretical value for a sudden expansion which is given in Reference 2 as

$$\eta = \frac{2}{1 + A_2/A_1}$$

. For the area ratio tested this gives an efficiency of 47.7% and although this figure is theoretical it gives some indication of the adverse effect of profile (6) on diffuser efficiency.

5.3 Tests on split diffusers

The three diffusers for which splitters were available were tested in the same manner as the plain diffusers, but the estimation of efficiency was complicated by differences in the static pressures occurring at the inner and outer walls of the diffuser throats. These static pressure differences varied with total head distribution, e.g., profile (1) and the 8.5° diffuser gave $(p_i - p_o)/\frac{1}{2}\rho\bar{V}_1^2 = 0.20$ whereas profile (2) with the same value of θ gave - 0.23. For any one profile, changes in θ also produced changes in static pressure, for example when θ was increased from 8.5° to 15° the static pressure difference corresponding to profile (1) decreased from $(p_i - p_o)/\frac{1}{2}\rho\bar{V}_1^2 = 0.20$ to 0.08.

The distribution of total head at inlet was however not affected by changes in θ. In the plain diffuser tests, where the wall static pressures were uniform, the mass flow calculated by integrating the velocity distribution at the throat gave good agreement with that measured at the orifice plate and thereby indicated reasonable uniformity in the circumferential distribution of total head. For the split diffuser tests a similar check, based on the assumption of a linear static pressure gradient across the throat gave differences in mass flow of up to 8% and so rendered the assumption regarding static pressure distribution invalid. In view of the unknown nature of the static pressure distribution across the throat a mean value of static pressure was estimated for each test by a process of trial and error using the orifice plate measurement of mass flow and the measured distribution of total head.

This static pressure and its accompanying velocity profile are those which would exist if static pressure were uniform, total head distribution identical with the measured distribution, and mass flow identical with the value measured by the orifice plate. Such conditions might be expected to obtain at some point sufficiently far upstream of the throat for the static pressure to be uninfluenced by the radial flows which are evidently occurring in the vicinity of the diffuser splitter.

Velocity profiles and diffuser efficiencies based on this 'equivalent' throat static pressure are shown in Figures 4 and 5. The results of a uniform profile test from Reference 1 are included in Figure 4 and appear to indicate much higher efficiencies than were obtained with profile (1). The efficiencies of Reference 1 were, however, based on total head traverses and the measurement of the outer wall static alone, and recalculation of the results for profile (1) on the same basis gave comparable efficiencies. The results based on 'equivalent' throat static pressure are believed to be more realistic, and comparison of Figures 4 and 2 shows that for a uniform inlet velocity profile a mean diameter splitter reduced efficiency at all values of θ up to 15°. This contradicts the conclusion made in Reference 1 that splitters improve efficiency at values of θ greater than 10°. The further conclusions in Reference 1 regarding the stabilising effect of splitters for values of θ greater than 10° is not affected.

Comparing the curve of 'nominal' efficiency for profile (1) with the results from Reference 1, profile (1) with its comparatively thin boundary layer appears to give slightly improved efficiency at $\theta = 15°$, but the effect is less marked than for the plain diffuser tests.

5.4 Discussion of results

In the preceding paragraphs the results for particular profiles have been described. These results have been scrutinised in various ways in an attempt to deduce the particular features of the profiles which dominate diffuser performance.

(a) Plain diffusers

It has been shown (Ref. 2) that a fully developed turbulent inlet profile gives less efficient diffusion than a completely uniform inlet profile. This reduction in efficiency can be attributed to the lower air velocities which are found near the walls in the case of the fully developed profile, this slowly moving air being more susceptible to breakaway under the action of the adverse pressure gradient than the higher velocity air of the uniform inlet profile. In analysing the six test profiles on this basis, the local wall velocity Vw was measured at a distance of 0.05 in. from the wall, this velocity being measured at the wall giving the lowest value, i.e. the wall at which breakaway is most likely to occur. Curves of η v $Vw/\bar{V}$ are shown in Figure 6 and show that test profiles (1), (2) and (4) form a group of high Vw's while profiles (3), (5) and (6) have low Vw's.

Clearly within each of these groups some further characteristic of the inlet profiles is having a pronounced effect on diffusion efficiency.

A second criterion of inlet profile was selected, namely the degree of symmetry of flow about the mean diameter expressed as $\overline{V_i}/\overline{V}$ where $\overline{V_i}$ represents the mean velocity in the inner half of the annulus and $\overline{V}$ is the overall mean velocity at the diffuser inlet. Curves of η v $\overline{V_i}/\overline{V}$ are shown in Figure 7A, the full lines referring to the high Vw profiles, and the dotted lines connecting points of low Vw. Therefore the vertical displacement between the full and dotted lines for any value of θ gives a measure of the effect of $Vw/\overline{V}$ on efficiency. It is clear that for $\theta = 6.5°$ diffuser efficiency is solely dependent on $\overline{V_i}/\overline{V}$ but as θ increases the effect of $Vw/\overline{V}$ becomes more evident until when $\theta = 15°$, a reduction in $Vw/\overline{V}$ from 0.98 to 0.78 lowers the efficiency by about 7%

(b) Split diffusers

A similar analysis has been made of the split diffuser tests and curves of η v $\overline{V_i}/\overline{V}$ are shown in Figure 7B. Clearly the effect of local wall velocity $Vw/\overline{V}$ is greatly reduced in comparison with the plain diffusers.

Optimum efficiencies are all reduced due to the additional friction loss introduced by the splitter, but efficiency is much less sensitive to symmetry of profile with the result that at $\overline{V_i}/\overline{V} = 1.20$ the split diffuser is more efficient than the plain diffuser.

In considering these results the geometry of the split diffuser should be borne in mind. The overall area ratio is 3.19 but the axial splitter provides two diffuser passages, an inner one of area ratio 2.67 and an outer one of area ratio 3.67.

Assuming the same efficiency for each section (true for small values of θ) it can be shown that a velocity profile having $\overline{V_i}/\overline{V} = 1.022$ will give a uniform static pressure in the diffuser throats. As θ increases, the efficiency of the larger area ratio section will tend to drop relative to that of the other diffusion passage and the velocity profile for uniform static pressure in the throats will approach the value of $\overline{V_i}/\overline{V} = 1.00$.

The curves in Figure 7B indicate that the condition of uniform static pressure corresponds to optimum efficiency, and therefore when inlet conditions are uniform, i.e. $\overline{V_i}/\overline{V} = 1.00$, the most efficient split diffuser will be one having equal area ratios in each section. For values of θ greater than 15° such a design should give an efficiency comparable to that of a plain diffuser and should cope efficiently with a much wider range of inlet velocity profiles.

6.0 Conclusions

Plain diffusers

1. A small angle diffuser ($\theta = 6.5°$) is capable of 80% efficiency or more over a wide range of inlet velocity distributions, the actual value of efficiency being dependent upon the degree of symmetry of the flow at inlet.

2. As the diffuser angle is increased, the efficiency becomes more sensitive to inlet conditions, being reduced by either non-symmetry

of flow or by the presence of a low velocity region near one of the walls.

Split diffusers

3. The introduction of a splitter, while improving stability (Ref. 1), reduces efficiency with uniform inlet velocity profiles for all diffuser angles less than 15°.

4. Splitting of the 10° and 15° diffusers improves efficiency for non-uniform profiles, but imposes pronounced static pressure gradients across the throat. In the application in which a split annular diffuser follows an axial compressor such pressure gradients might prove undesirable.

5. For the application in which the inlet distribution, although non-uniform, is likely to remain constant, a high angle split diffuser with individual area ratios designed to give a uniform static pressure distribution across the throat should give better efficiency than a longer (i.e., smaller angle) plain diffuser.

Acknowledgment

The author is indebted to Messrs. G.G.P. Rooker and A.I. Lloyd who conducted a large part of the experimental work described in this paper.

Notation

A_1	=	inlet area
A_2	=	outlet area
p_1	=	mean inlet static pressure
p_2	=	mean outlet static pressure
p_i	=	inlet static pressure at I.D.
p_o	=	inlet static pressure at O.D.
V_{max}	=	maximum velocity at inlet
$\bar{V}_1$	=	mean velocity at inlet
V_w	=	velocity at inlet 0.05 in. from wall
$\bar{V}_i$	=	mean velocity over inner half of annulus
θ	=	diffuser angle
η	=	diffuser efficiency
R_e	=	Reynolds number

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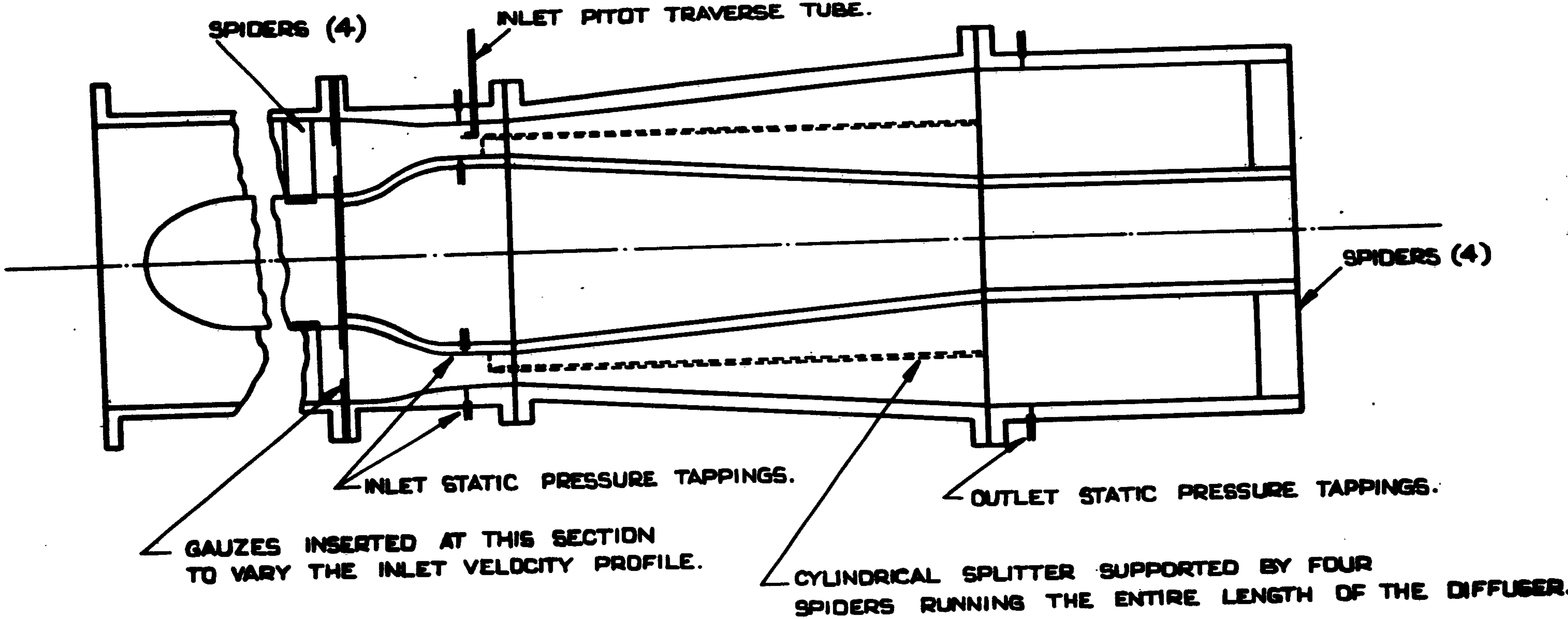

DIAGRAM OF DIFFUSER RIG.

SHOWING 8·5° DIFFUSER

SCALE :- 5:1

FIG. 1.

FIG. 2

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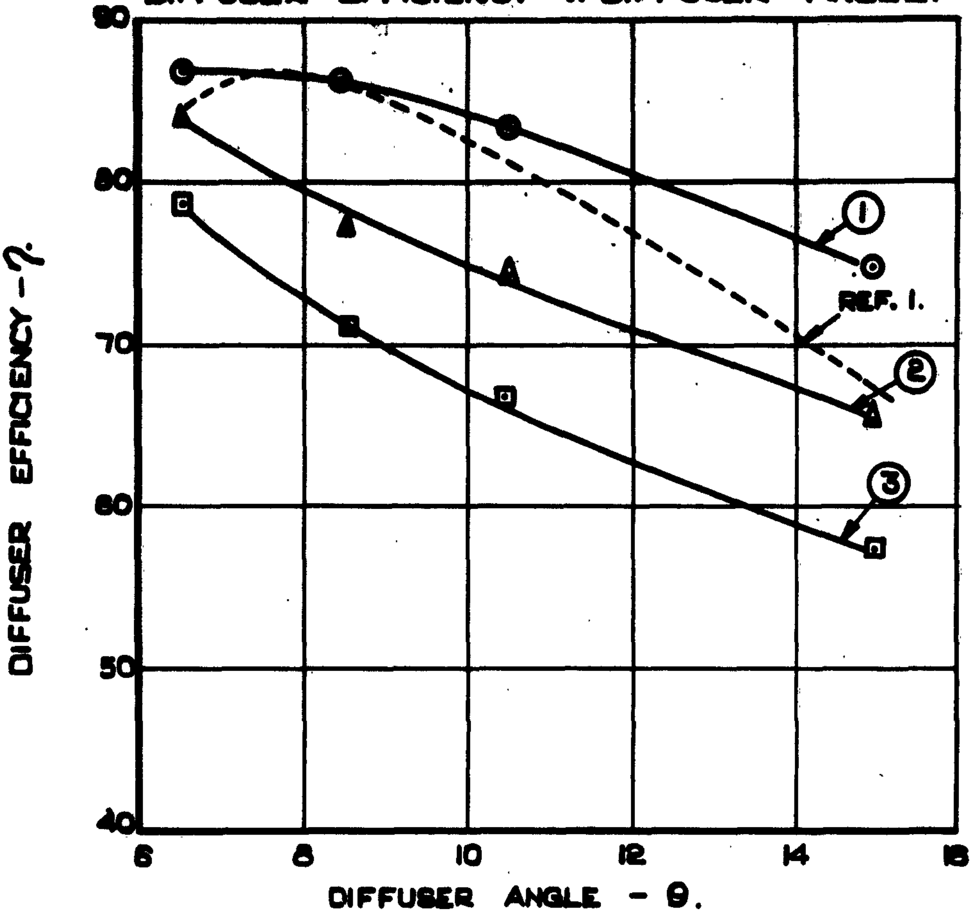
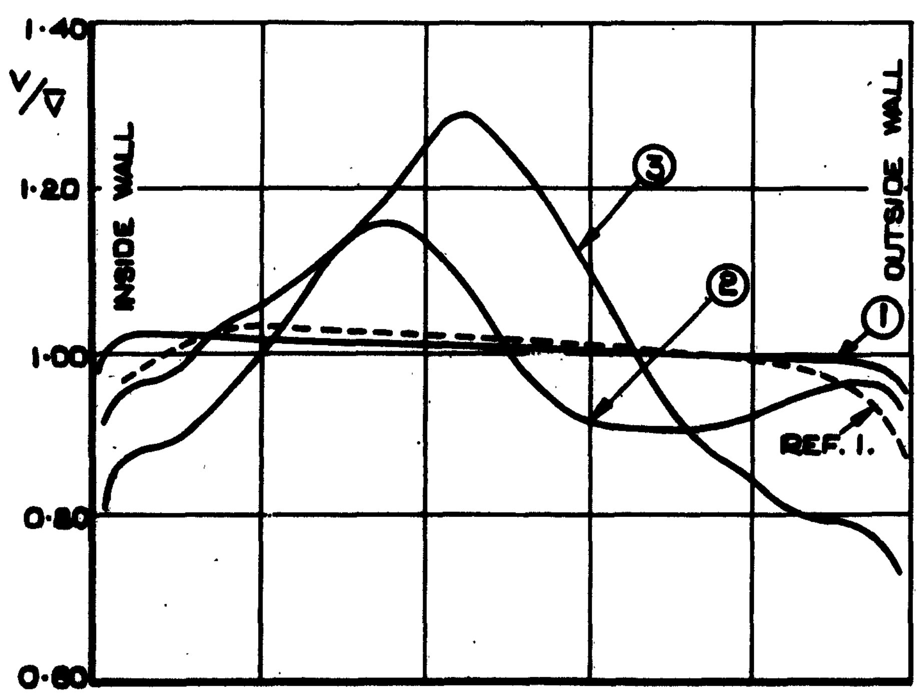
DIFFUSER EFFICIENCY V. DIFFUSER ANGLE.

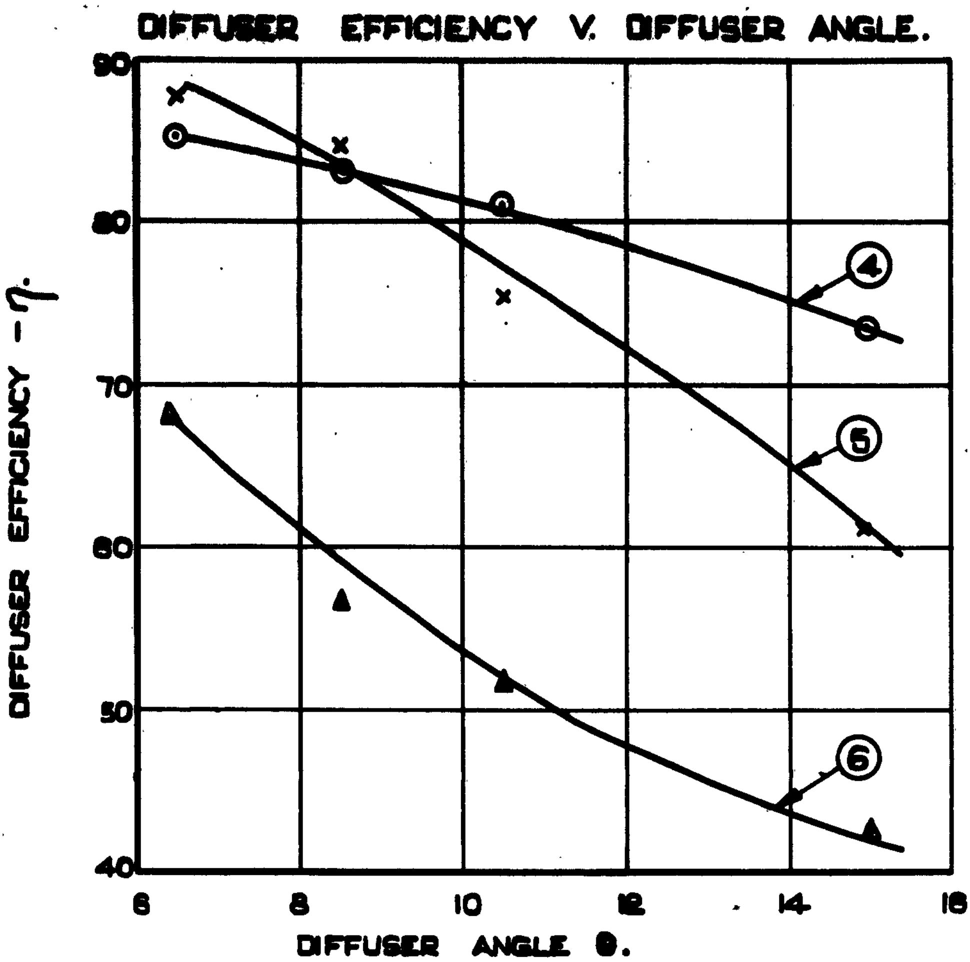


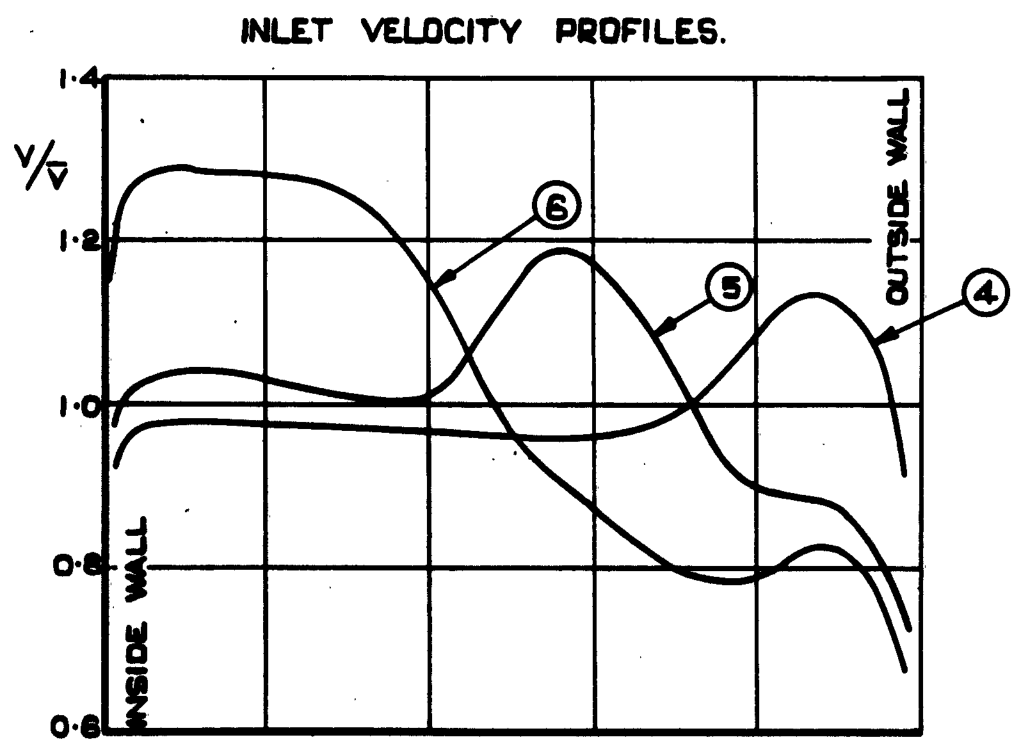
INLET VELOCITY PROFILES.



PLAIN DIFFUSER TESTS
EFFECT OF VELOCITY PROFILE ON EFFICIENCY.

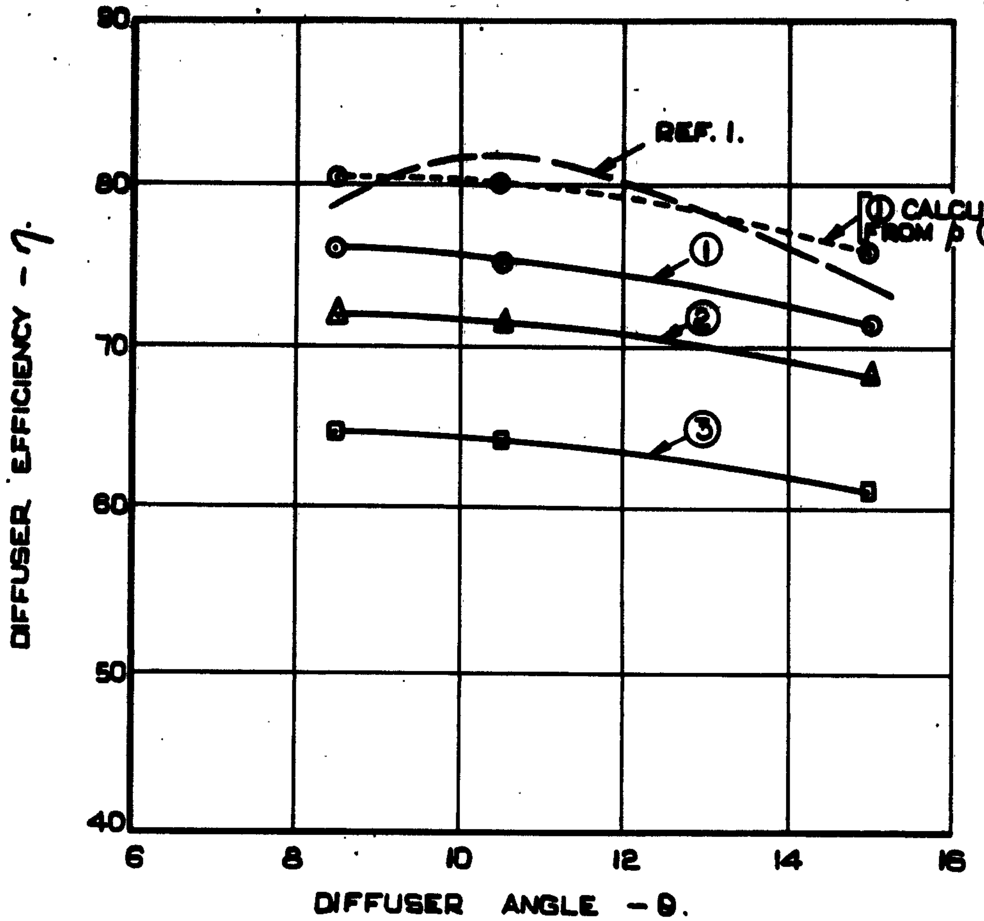
FIG. 3



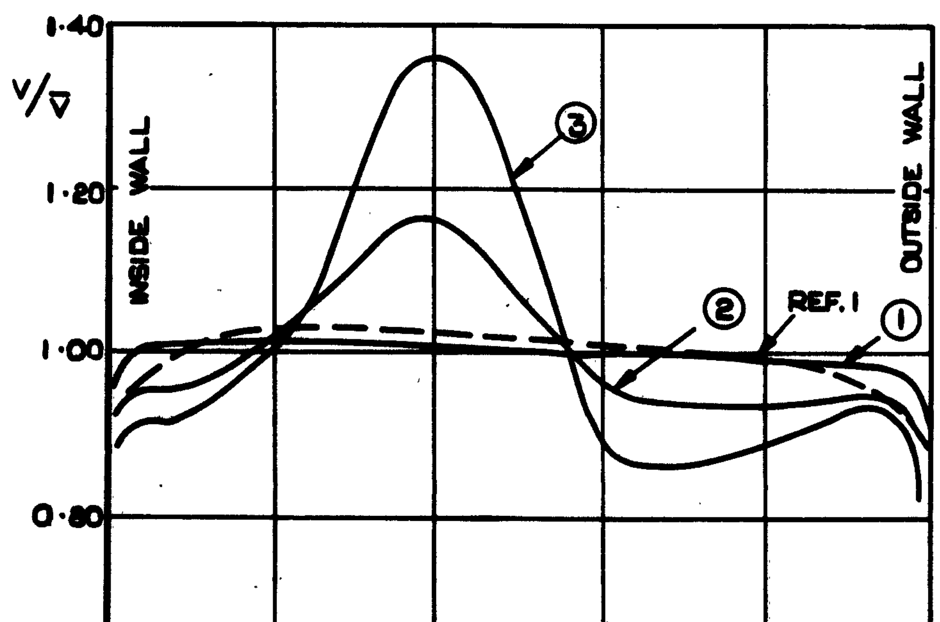


PLAIN DIFFUSER TESTS
EFFECT OF POSITION OF PEAK VELOCITY ON EFFICIENCY.

DIFFUSER EFFICIENCY V. DIFFUSER ANGLE.



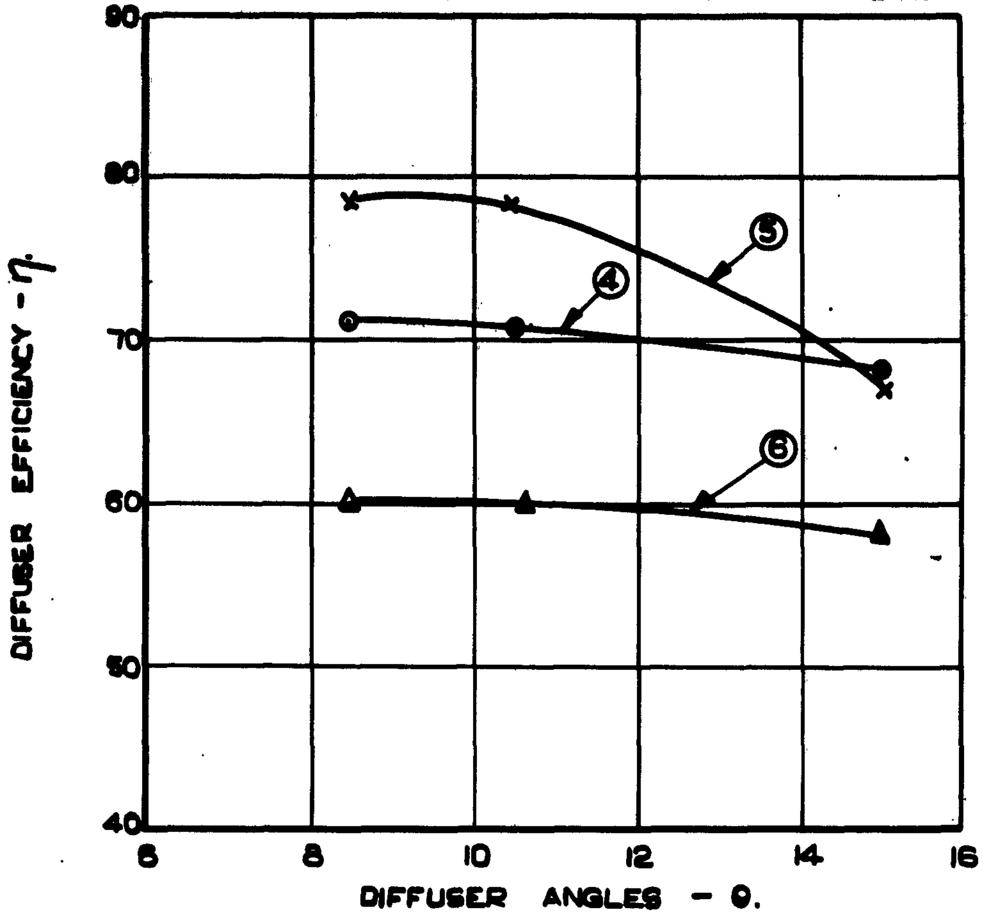
INLET VELOCITY PROFILES.



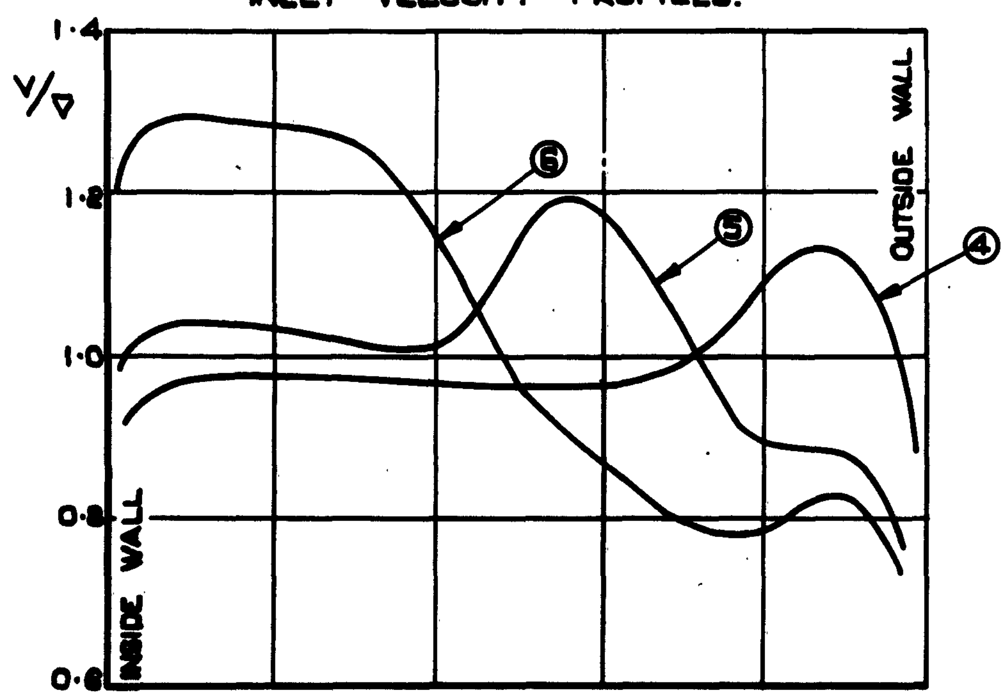
SPLIT DIFFUSER TESTS
EFFECT OF VELOCITY PROFILE ON EFFICIENCY.

FIG. 5

DIFFUSER EFFICIENCY V. DIFFUSER ANGLE.

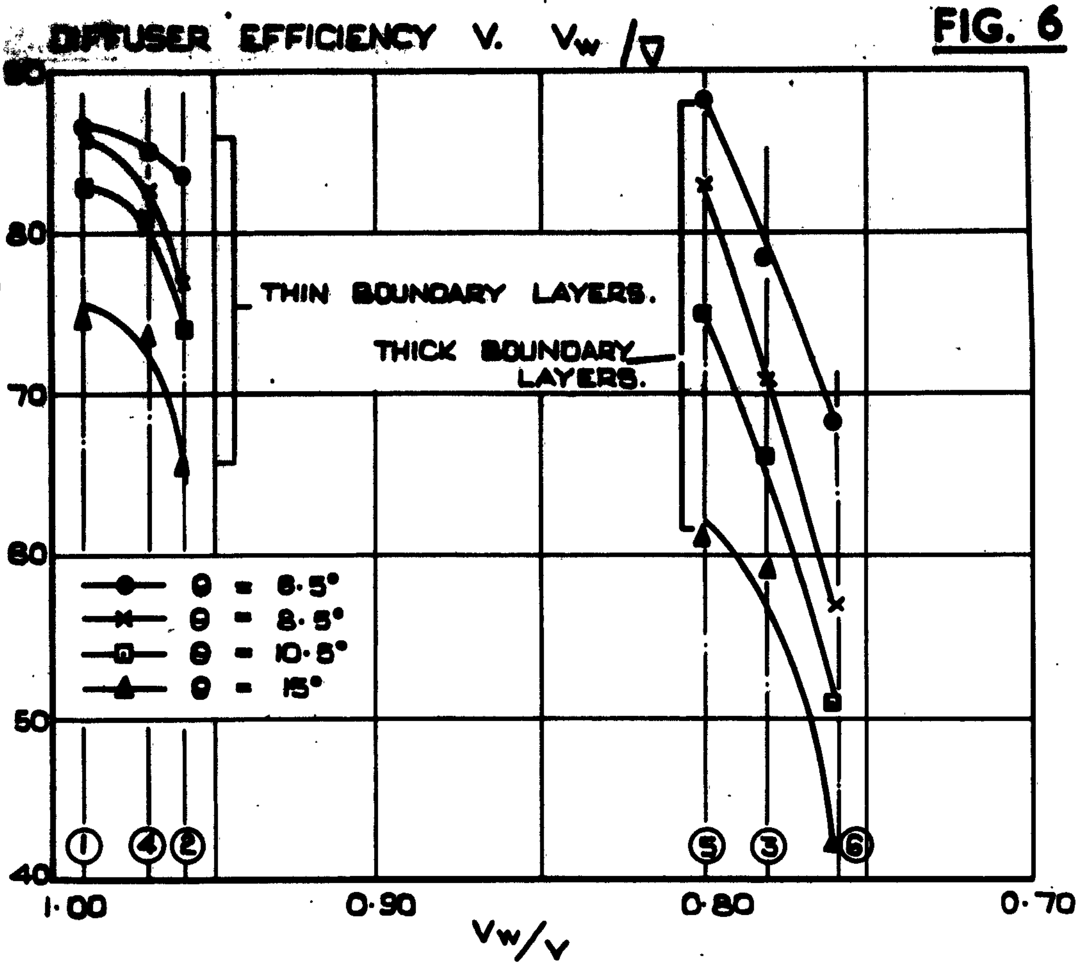


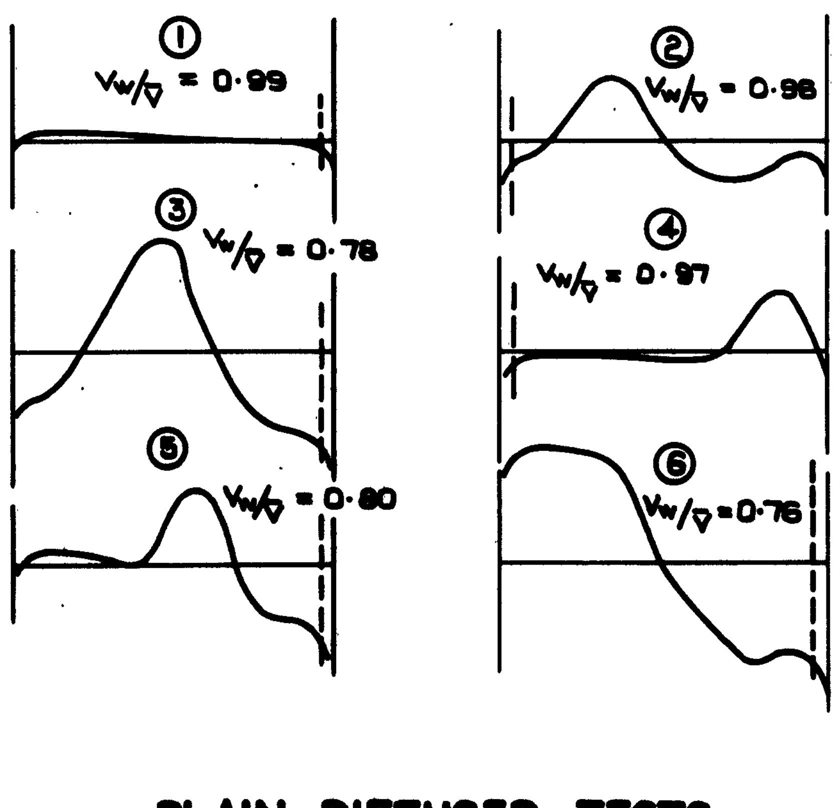
INLET VELOCITY PROFILES.



SPLIT DIFFUSER TESTS
EFFECT OF POSITION OF PEAK VELOCITY ON EFFICIENCY.

FIG. 6





PLAIN DIFFUSER TESTS THE EFFECT OF LOCAL WALL VELOCITY ON EFFICIENCY.

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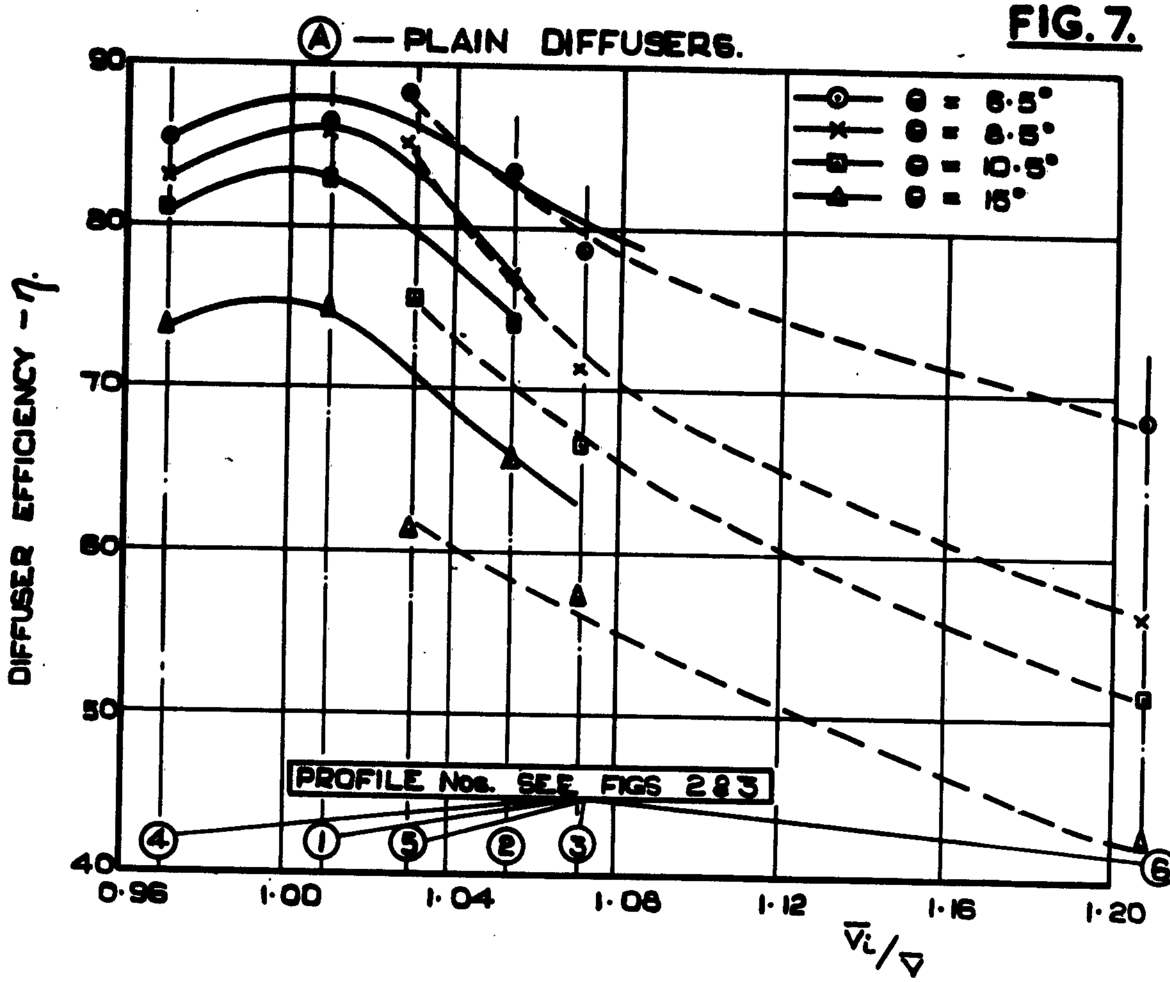
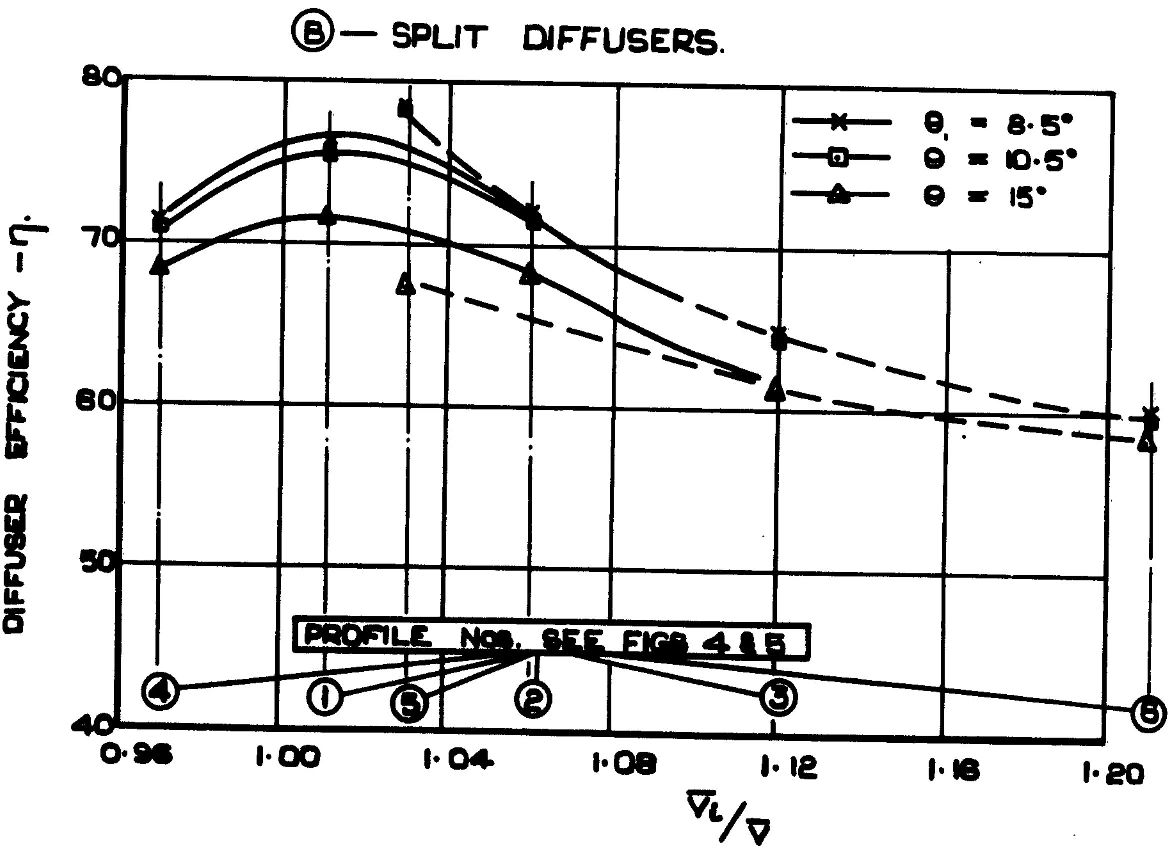
FIG. 7.





DIFFUSER EFFICIENCY V. $\bar{V}_i/\bar{V}$

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